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THE DESIGN AND OPERATION OF AN INFRARED SIMULATOR FOR TESTING OF THE SHUTTLE RADIATOR SYSTEM

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ABSTRACT

An infrared simulation system was designed and installed in chamber A of the NASA Johnson Space Center in support of the development of the Space Shuttle Modular Radiator System. Various types of simulator designs were considered for conditioning of the test article during thermal-vacuum testing. This paper discusses the design and operation of fluid controlled panels to simulate the incident infrared flux which would be experienced by the Space Shuttle radiators during earth orbit. The thermal capabilities of the simulator are presented together with the reasons for the selection of this type of system.

INTRODUCTION

A series of tests were conducted in the Space Environment Simulation Laboratory chamber "A" at the NASA Johnson Space Center to determine parametric thermal performance of a candidate modular radiator system proposed for the space shuttle. The conventional heat sink and solar simulation systems of the chamber could not be used for these tests because of the size of the radiator system, and the requirement for simulating different heat loads simultaneously for various combinations of the eight radiator panels comprising the array. The major requirements for thermal simulation were as follows:

- o Heat flux simulation had to be provided by individually controlled modules for each of the eight radiators.
- o The infrared (IR) flux absorbed by each radiator had to be uniform and individually controllable in the range 20 to 150 Btu/ft^2 hr at an accuracy of 5 Btu/ft^2 hr.

- The average flux actually obtained from the IR simulator had to be within ±10 percent of the desired flux level.

- For cold soak conditions a flux level of approximately 5 Btu/ft^2 hr had to be provided.

- The accuracy of the actual flux had to be known within ±5 Btu/ft^2 hr.

- The flux at any point on each radiator had to be within ±10 percent of the average flux for that specific radiator.

- The original test plan outlined a series of steady state conditions within the flux range; orbital profile simulation was not required. However, requirements for dynamic orbital simulation were subsequently identified and these requirements were also accommodated by the system design described below.

DESIGN

IR Simulator System Selection

An extensive design study was conducted where several simulator concepts were evaluated to provide the above thermal simulation requirements. Among these were electrical strip heaters, quartz lamps, and fluid panels. Strip heaters of thin nichrome were available from a previous test program and could be assembled to provide individually controlled zones. The main disadvantage of this design was that some sort of rotatable or retractable mechanism had to be provided for each panel to meet the cold soak and low flux profile requirements due to excessive blockage of the chamber cold wall. Quartz lamps were not selected primarily due to the difficulty in providing the absorbed flux uniformity on the large 6- by 12-feet test articles. In addition an infrared simulator designed with quartz lamps or strip heaters would require full chamber cold walls during the testing. The initial cost of the fluid panels was high, but the relatively simple thermal analysis and accuracy of providing known flux levels on a real-time basis more than offset the initial higher cost. The fluid panels had the additional advantage of being cold soaked rapidly with LN_2, eliminating the slow cooling by radiation to the chamber cold sink. The thermal simulation system that was selected was a set of eight individually controlled fluid conditioned panels. The overall rectangular dimensions of the panels were the same as the test article radiators, 6- by 12-feet. Each simulator panel was installed directly below a test radiator (see fig. 1). The majority of the testing was conducted by enclosing the test radiator and the simulator panels in radiation shields, thereby eleminating the need for full chamber cold walls. Each panel had two independent flow systems, one for Freon-11 and a second for gaseous or liquid nitrogen. The Freon-11 was used to provide controlled flux levels between 20 and 150 Btu/ft^2 hr, and the LN_2 was used for the 5 Btu/ft^2 hr condition. Controlled fluxes between 5 and 20 Btu/ft^2 hr could not be obtained due to the -168° F freezing temperature of Freon-11. The Freon-11 heating system had the capability of both manual or closed loop computer control. The hot gaseous nitrogen system was used primarily to warm the panels from LN_2 temperature to above -130° F at which point the Freon-11 system could again be used. In addition it served as a backup for simulation of higher flux levels if problems occurred with the Freon-11 system. Eight identical systems were installed with a unitized control console and a common Freon-11 reservoir.

Thermal Considerations

The fluid panels were designed to have a maximum of 5° F drop from the inlet to the outlet manifolds and a 3° F differential between the fluid channel centerlines when the panel conditioning was provided by Freon-11. The thermal calculations were based on the assumption that the heat transfer from the fluid is exchanged with the panel at the welded joints (see following section, "Mechanical"). The 8-inch Freon-11 channel spacing was based on the calculations that the heat exchanged by the panel is from the midpoint between the fluid channels to an element and must pass through the element to the liquid. When view factors and emittances are considered the average temperature differentials between the Freon-11 channels was calculated to be 3° F and 15° F between the LN_2 channels. The Freon-11 flow rate calculated for a maximum of 5° F temperature differential along the tube was based on an heat exchange between the radiator and simulator panel of 150 Btu/ ft^2 hr. Using a panel size of 6- by 12-feet and the specific heat of Freon-11 being 0.21 Btu/lb °F, the minimum Freon flow required was 10,300 lb/hr or 14 gal/min.

In order to provide a surface with a high emissivity, the simulator panel surface on the opposite side of the tubing was coated with approximately 3 mils of 3M 410C Nextel black matte paint. A special coating procedure was required to withstand the temperature extremes (-320° F to +130° F) that would be encountered during the testing. The aluminum surface was first abraded with sandpaper and then washed with Freon MF. After drying, a coating of 3M 901-P1 primer was applied and the 3M 410C paint followed within a few minutes by spraying. The large panels were cured at 250° F for 24 hours by flowing steam through the tubes. This coating did not deteriorate even after several hundred hours of thermal-vacuum testing and also provided as high as 216 Btu/ft^2 hr flux levels (under limited radiator temperature conditions) without exceeding the maximum Freon-11 system operating pressures.

The incident flux level as a function of the simulator panel temperature is presented in figure 2. Also shown is the approximate flux absorbed by the radiator (emittance 0.85). During actual testing the simulator panel temperatures had to be adjusted to compensate for the reflected energy between the panel and the radiator.

<u>Mechanical</u>

The preceding thermal considerations resulted in the mechanical design illustrated in figure 3 and figure 4. An aluminum plate 0.187 inches thick was chosen to provide the conductive path, with the Freon tubes spaced on eight-inch centers. One-inch square tubing was selected for the flow paths. Staggered full fillet welds were used to attach the tubing to the plate, which also provided a good conductive path to the plate. Attached at every second tube was another of the same size to form a separate GN_2/LN_2 flow path. Placing one tube on the other also increased the moment of inertia so that the panel was stiff enough to be supported at only two places with no noticable sag. The 1-inch tubes were welded to 1-1/4-inch square tube manifolds on each end. The manifolds were sized to provide equal flow in each of the fluid channels.

Simple aluminum angle tables were used to support the test radiator-simulator panel assembly (see fig. 1). Isolation brackets were built which supported the shuttle radiators a uniform 6 inches above the thermal simulator panels. The whole assembly was wrapped with radiation shielding and operated without the necessity of cooling the entire chamber to LN_2 temperature for some test modes. For certain test modes, the radiation shields were also placed at an angle and used with full LN_2 cold walls to simulate blockage by the shuttle cargo bay doors.

The radiator and simulator panel pairs were arranged in the chamber as shown in figure 5. This panel arrangement provided a close simulation of the Shuttle cargo bay door plumbing installation. Aluminum tubing (6061-T6) of 1-1/4-inch diameter was used for the Freon-11 and the GN_2 /LN_2 plumbing. All in chamber lines were welded and helium leak-checked prior to the installation of two layers of aluminized Kapton insulation.

Heat Supply

To supply the heat output of 10,800 Btu/hr a power rate of 3.2 kilowatts was required. The mass of the panel, interconnecting plumbing and fluid was calculated to be approximately 600 pounds. Assuming an average specific heat of 0.22 Btu/lb °F, the transient power constant was 2.3 KW min/°F. A 9 KW Calrod heater (22 watts/in^2) and a 5 horsepower, 50 psi centrifugal pump was installed for the Freon-11 circulating system. Assuming about 1/4 of the motor power going into the fluid, approximately 10 kilowatts of heat was available. This provided a transient of 3° F/min rise in the system temperature.

Heat Rejection

Cooling of the Freon-11 was accomplished by using a Freon/LN_2 heat exchanger. The cooling requirements for average steady-state conditions were 10,800 Btu/hr for heat input from the test radiators, plus pump cooling, heat leaks in the lines, and some trim heat to ensure computer control. This gave an approximate fixed heat load of 18,000 Btu/hr. The cooling required during transient conditions was about 40,000 Btu/hr. The total maximum cooling load was 58,000 Btu/hr.

The Freon/LN_2 heat exchanger was sized to provide the cooling with an LN_2 flow of approximately 1 gallon per minute.

Freon Fluid System

The temperature of the simulator panels was controlled by its own independent fluid system; a schematic illustrating the major components is illustrated in figure 6. A single Freon storage tank supplied the Freon-11 to the zones. A separate Freon drain tank and transfer system (not shown in the figure) was used to drain the system prior to initiating LN_2 to the

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panels. The independent circulating and control system for each panel provided flexibility in controlling the panel temperatures at different levels. An alternate approach which was considered and rejected consisted of a central large capacity pumping system with the necessary controls to blend the hot and cold fluid streams. This system could not provide the required flexibility without the complexity of an elaborate control system. A 20 gpm, 50 psig discharge pressure centrifugal pump with standard controls was used to circulate the temperature controlled Freon-11 through the panels. A 60 psig blanket pressure was maintained on the storage tank to provide sufficient net positive suction head for the high temperature operation. Aluminum tubing 1-1/4inch in diameter was used for the liquid flow paths. The Freon-11 temperature was controlled with a 9 kw Calrod heater and a 100,000 Btu/hr, LN_2 /Freon heat exchanger. A needle metering valve was modified for cryogenic application and was used for manual control of the LN_2 to the heat exchanger. The electrical heater could be controlled manually or by automatic computer control in increments of approximately 145 watts. During most operations the system was set to always require some trim heat since the final temperature control of the panel was maintained by the facility computer.

Computer Control

The 9 kw heaters were controlled by a facility computer. The software computer program used previously for an electrical IR simulator was modified to permit either open- or closed-loop control of the heater to provide a predetermined simulator panel temperature. The software package provided considerable flexibility through real-time input options which included the following:

- Selection of open or closed-loop control.
- Selection of the time interval between power control signal updates.
- Definition of thermocouples to be averaged for closed-loop temperature control.
- Selection of primary and secondary control modes and cathode-ray tube display of the average panel temperatures.

The power controllers were silicon-controlled rectifiers capable of producing a 117-volt rectified alternating current. The computer program generated a 6-bit binary power input signal to each control which resulted in a heater control of approximately 145 watt increments.

The IR simulator control sensors were copper/constantan thermocouples, which were installed on the tube side of the panel. The distribution of the thermocouples on the simulator panels is shown in figure 4. (The location dimensions are in inches.) A coded measurement numbering system was incorporated, the second digit of the number is the simulator panel number, while the last two digits are the thermocouple number.

The control program read the desired average panel temperature and compared this value to the actual temperature for the current update interval. The program then corrected the command to the power controllers to cause the measured temperature to approach the desired value. The control system options enabled the thermal engineer to provide the precise thermal simulation required for the specific test mode and to make real-time corrections as required. A comparison of the desired and actual flux profiles is shown in figure 7.

LN_2/GN_2 System

The unitized control console (see fig. 8) with the valving, flow and pressure indicators provided controlled closed-loop and single-pass LN_2 flow from the facility system into the panels. Alternate valving provided heated GN_2 through the same piping. Each loop was independent of the others except for the common facility supply. The heated gas was used primarily to bring the panel temperature above the freezing point of Freon before reloading the Freon loop. The heated GN_2 was also used as a redundant system for flux levels up to about 130 Btu/ft^2 hr.

Thermal Calculation Technique

A mathematical model of the radiator-simulator panel assembly was derived by assigning 12 thermal nodes to the radiator, one node to the IR simulator panel and four nodes to the radiation shields enclosing the perimeter of the assembly (see fig. 9). Based on these nodes view factors were calculated. A desktop calculator was programed using these view factors, the emissivity of both the radiator and the IR simulator panels, and the node temperatures to calculate the average absorbed flux and that absorbed by each node. Pretest predicted radiator temperatures were entered and iterative panel temperatures were tried until one was selected which would provide the desired absorbed radiator flux. During the test phase the measured radiator and panel temperatures were used to calculate the actual absorbed flux into the radiator. After some testing the thermal engineers became familiar with the system operation, and the number of radiator nodes were reduced from twelve to four without any loss of accuracy in the calculations. For final data analysis of the absorbed flux, only one node was used. Selected thermocouples were used to meld the flux distribution into an accurate average.

SYSTEM OPERATION

The first series of tests were conducted with the radiator and simulator panels completely enclosed by radiation shields. (See fig. 1.) The radiation shields were multilayer super-insulation blankets. This configuration provided insulation on one side of the radiator simulating operation in the "single-sided" radiation mode. For the "two-sided" radiation test configuration the blankets were removed so that the top side of the radiators would reject heat to the chamber cold walls. A third series of tests were conducted by positioning the blankets at an angle above the radiator. This configuration simulated "two-sided" radiation with some blockage by the Shuttle cargo bay doors. Initially the theoretical radiator temperature patterns were used to determine the simulator panel temperatures for the desired flux level. The thermal simulation using these values provided a flux level that was approximately 5 percent high. Between test series the emittance of the test radiator coating was measured and was found to be higher than had been expected. The calculator program was corrected for the new emittance value and the simulated flux levels agreed with the analytical predictions. The fluid system had been initially designed to the requirement for steady-state levels of IR simulation. After the system was fabricated, simulation of dynamic orbital profiles were requested. When this was first attempted, Freon-11 was left in the panels and was frozen by the LN_2 while attempting simulation of the cold side of the orbit. In order to initiate Freon-11 flow again for the warmer environment simulation, hot GN_2 was introduced in the LN_2/GN_2 manifold in an attempt to thaw the frozen Freon-11. Nonuniform heating of the Freon-11 channels caused expansion of the liquid between frozen sections resulting in excessive pressure build-up and the subsequent rupture of the Freon-11 manifold. This problem was solved during subsequent testing by draining the Freon-11 loop before introducing LN_2.

The calculator programs developed for this test proved to be indispensable for determining panel set-point temperatures and for quick-data reduction to verify that proper flux levels were being provided. The IR system met all the original design requirements plus the added requirement for orbital cycles. It also was used to provide fluxes as high as 175 Btu/ft² hr. Coincidentally, the thermal mass of the system was such that the heating and cooling rates very closely met the 90-minute earth orbital profile. The system was not originally designed for this requirement. The closed loop computer control of the heater proved especially effective during steady state flux conditions. It allowed the test engineer to maintain the final fluid temperature control after the initial approximate manual setting was made at the control console.

Figure 10 illustrates a typical case of providing both steady state and orbital cycle flux profiles. This graph was generated by a computer program available at JSC which calculated the average flux levels from the test data tapes and provided both tabulated data and graph plots.

Figure 11 shows a plot of the same time period for a different Shuttle radiator where the flux level was controlled by the GN_2/LN_2 loop after the Freon control loop had failed. Note the close similarity with figure 9 which demonstrated the back-up capability of the GN_2/LN_2 system.

A typical radiator flux level trace is presented in figure 7. These data show the desired absorbed steady state and cyclic flux levels compared to the actual flux levels. The slopes and the steady state drift were well within the design requirements indicating the excellent performance of the IR simulation system.

CONCLUSIONS

The fluid controlled IR simulation system was a good choice for this application because of the flexibility in providing desired flux levels over a wide range on a real-time basis. The system response time was very good in that a single zone could be changed from cold soak LN_2 temperatures to controlled warmer environment simulation with the Freon-11 loop in approximately 30 minutes. Approximately 2 hours were required for this sequence for all eight panels. The time required to change all eight panels from a controlled flux level using the Freon loop to stable LN_2 temperatures was approximately 90 minutes.

The programed desk calculator was an essential aid to the thermal engineer in determining the panel temperatures necessary to provide the absorbed infrared flux by the radiators. The data could be analyzed during the actual test sequence and rapid real-time adjustments could be made as required to provide the exact flux level. The simulator system exceeded the design requirements in all cases and in addition provided orbital flux simulation.

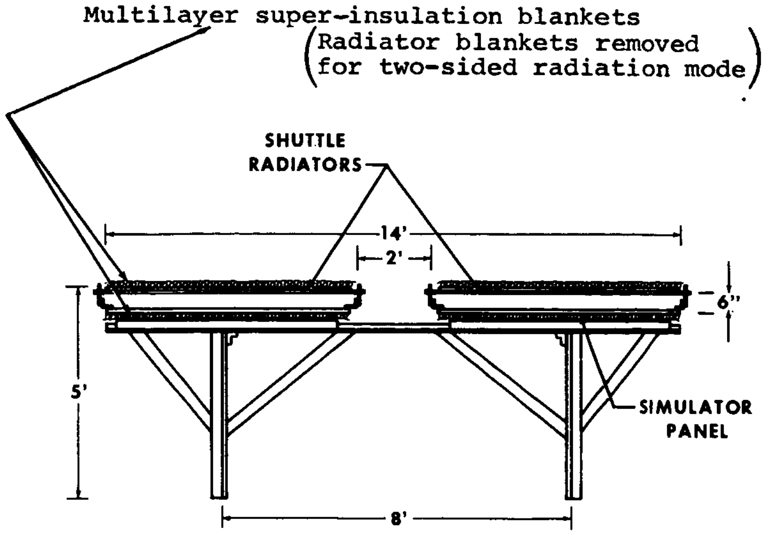

Fig. 1 - End-View of typical radiator-simulator pair installation

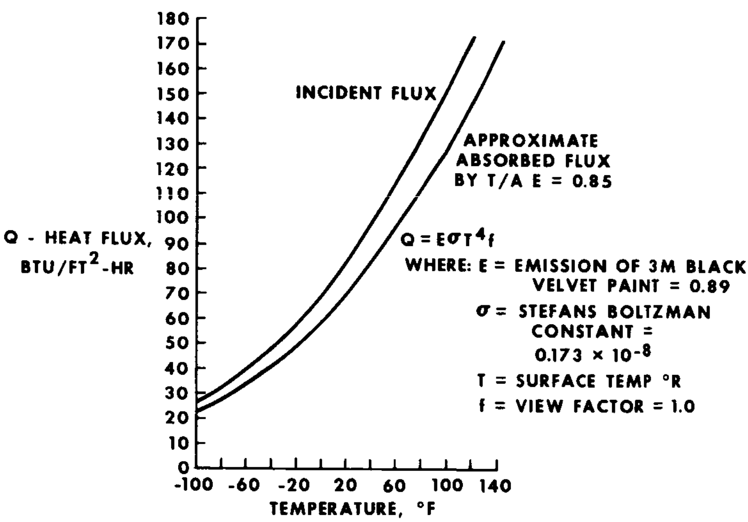

Fig. 2 - Simulator panel flux as a function of temperature

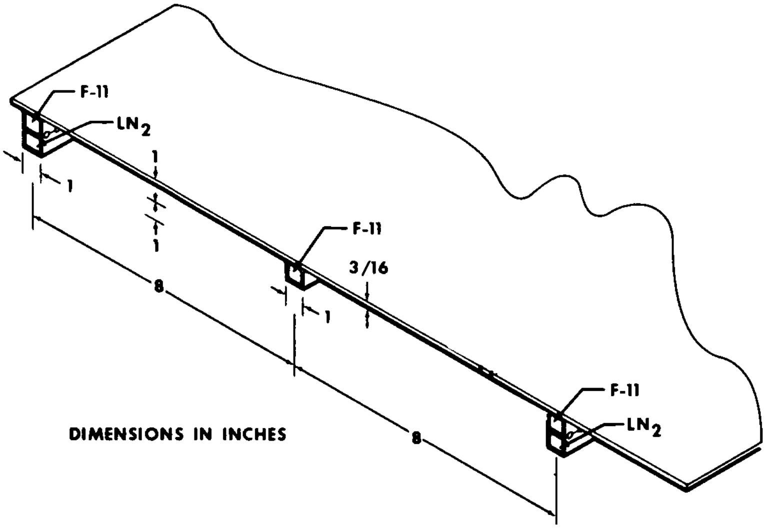

Fig. 3 - IR simulator panel fluid tube spacing

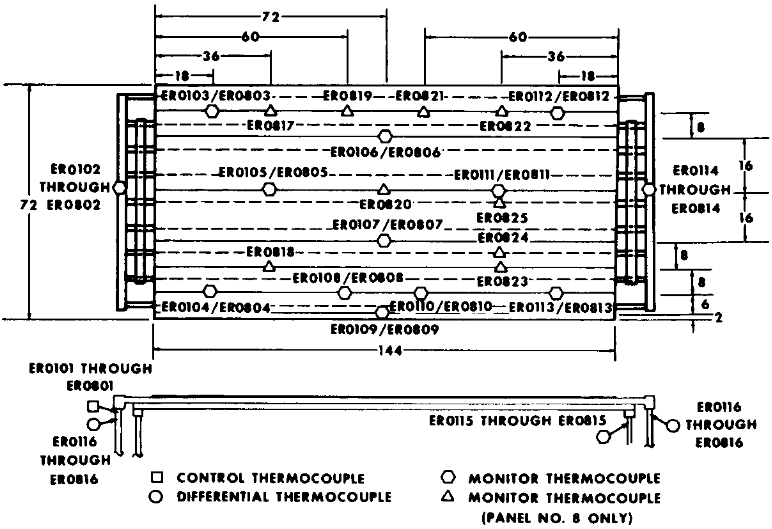

Fig. 4 - Piping and thermocouple distribution on simulator panel

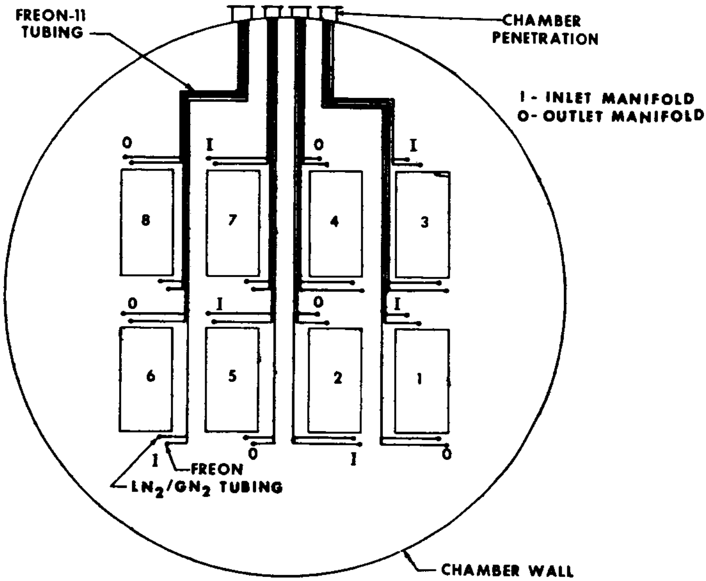

Fig. 5 - Chamber A installation of of radiator/simulator pairs

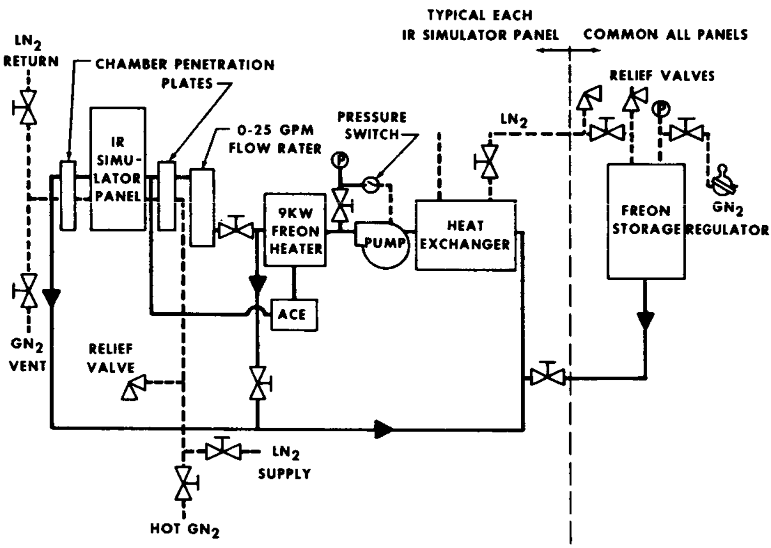

Fig. 6 - Schematic of individual fluid control system

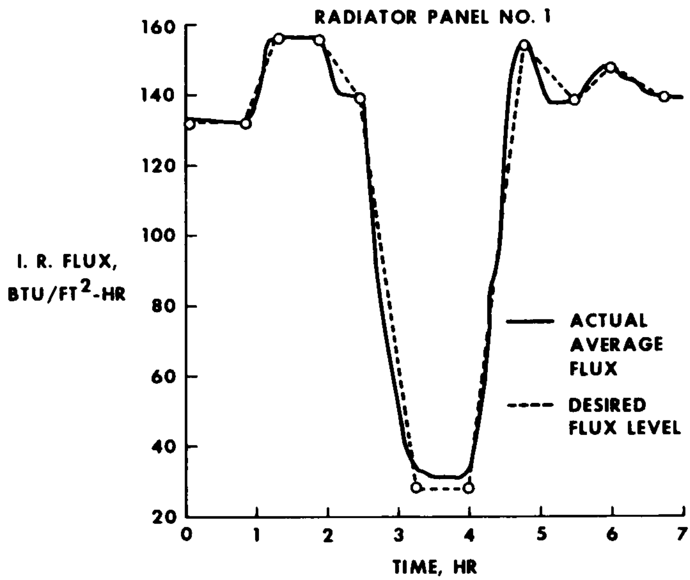

Fig. 7 - Typical simulator performance compared with desired profile

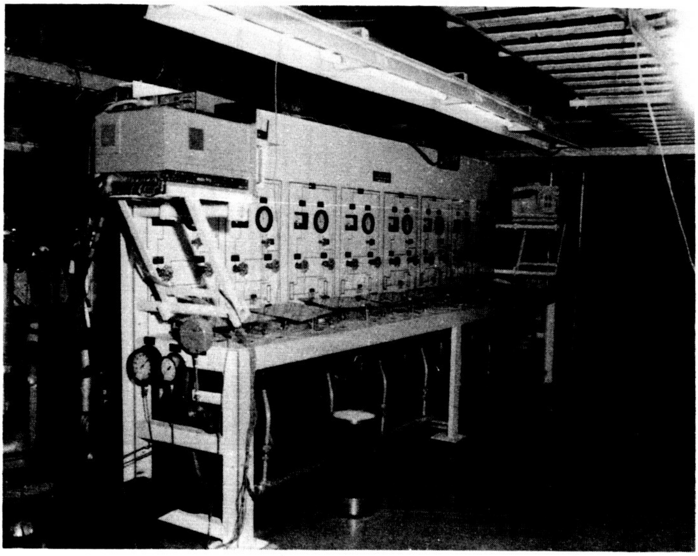

Fig. 8 - Unitized fluid control console

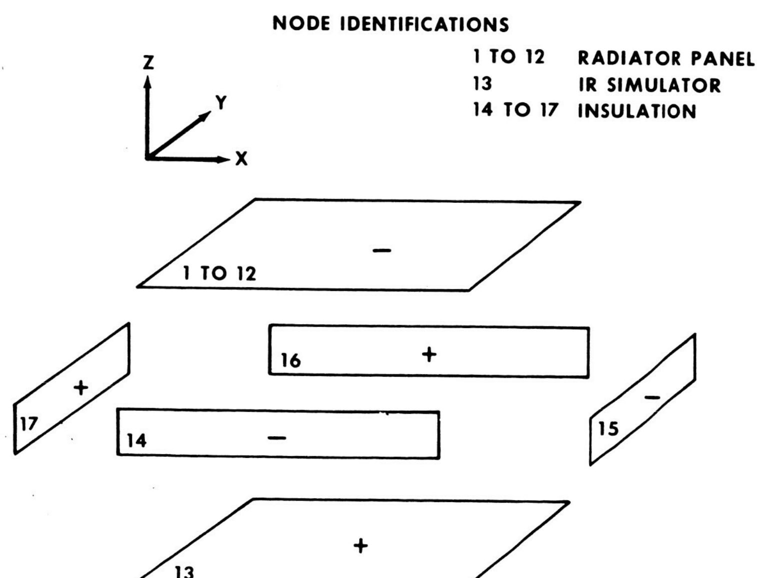

Fig. 9 - Mathematical model of radiator/simulator installation

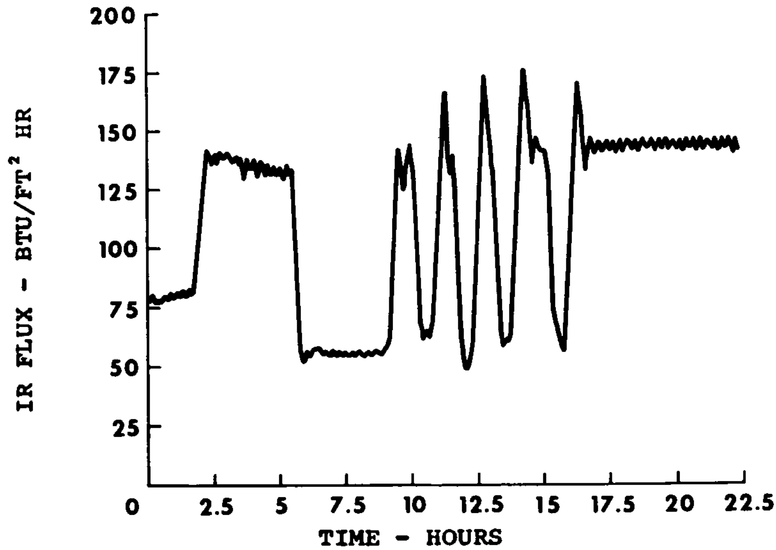

Fig. 10 - Simulator performance with Freon-11 operation

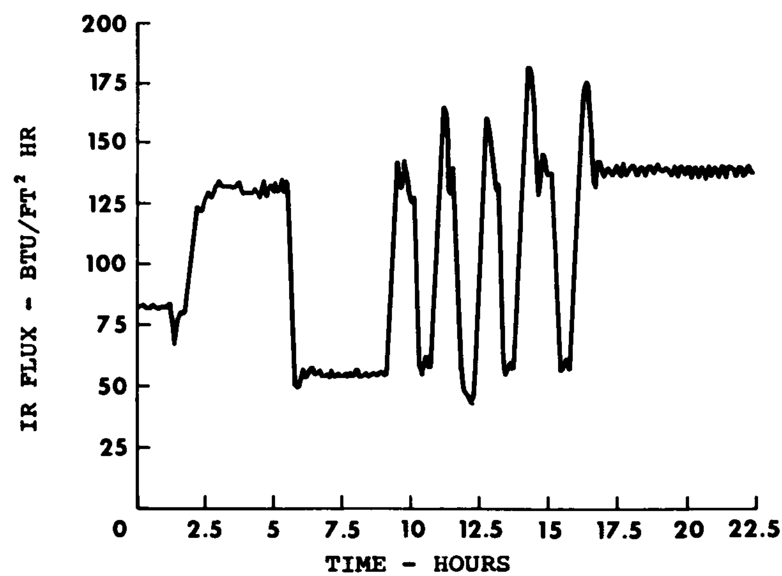

Fig. 11 - Simulator performance with GN_2/LN_2 operation